Axial Force Damper in a Linear Wave Energy Converter

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Abstract

This paper presents a study on the axial force damper in a wave energy converter. The converter itself consists of a linear generator placed on the ocean floor connected to a buoy at the ocean surface. As the development of the complete system goes forward, the economical perspective becomes more and more important. In order to ensure an economically viable alternative to the electric energy conversion, the costs associated with the use of materials have to be reduced while the survivability of the wave power unit has to be prolonged. The study presented in this paper increases the knowledge on axial forces and illustrates the damping system required to prevent the failure of the hull which houses the generator. The results are aimed to be utilized in the future design of wave energy converters and influence the choice of materials, the total costs and prolong the survivability of the wave energy converters in harsh wave climates.

Keywords

Wave Power; Axial Force; Ocean Energy

Introduction

Mankind has been utilizing the streaming water in rivers for a long time, both to perform mechanical work and later to convert the flowing kinetic and potential energy of the water to electrical energy. The energy in the oceans has not historically been extracted in large scale, even through it has a great potential. According to a recent estimation, the potential power production of total 1 TW over the globe (Falner et al., 1991), represents 41 % of the total electric energy conversion in the world during 2010 (Global Energy Statistical Year Book).

There are different conversion technologies that can be divided into different categories according to the operating principle: oscillating water column, overtopping devices and wave activated bodies. At the Swedish Centre for Renewable Electric Energy Conversion at Uppsala University, a system to utilize and convert wave energy to electrical energy has been designed. It falls in the category of wave activated bodies and consists of a direct driven linear generator installed at the seabed connected by a line to a point absorbing buoy, illustrated in Fig 1.

As the buoy moves with the waves, a relative motion between the translator, equipped with permanent magnets, and the cable-wound stator appears and an

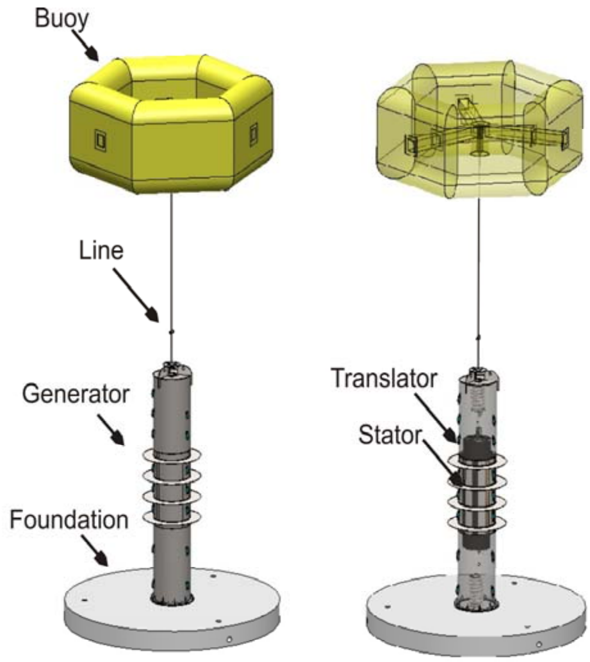


FIG. 1 ILLUSTRATION OF A WAVE ENERGY CONVERTER

induced voltage in the stator cables can be measured. A mechanical design without a gearbox or intermediate energy storage between the buoy and generator keeps the system simple which is believed to increase the lifetime of the device. Another advantage of this technology is its modularity. A wave power plant consists of an appropriate number of

units and, as the demand grows, more units can be added.

The Lysekil Wave Power Project has so far resulted in the construction and offshore installation of eight different wave energy converters at the Lysekil Research Site, located at a few kilometres south-west from Lysekil, a town at the Swedish west coast. The experimental site has an performable environment covering 10 WECs, two marine substations and 30 biology buoys. A wave measuring buoy, a *Waverider* installed since 2004, continuously measures the sea state and an observation tower equipped with a camera, located in the southern part of the site, is utilized for research purposes (Tyberg et al., 2009; Tyberg et al., 2011) and offers an overview of the experimental site. A detailed description of the wave energy project, the experimental work and main progress within the Lysekil Wave Power Project at the Lysekil Research Site is presented in (Lejerskog et al., 2011; Leijon et al., 2009).

The WEC technology developed at Uppsala University is challenging. An important goal during the development of a new wave energy converters is to reduce the amount of material, and thereby the cost, in the mechanical design. In order to reach this objective, the maximum stresses exposed to the generator have to be estimated in an accurate way. This paper develops a method to design a damping system of the axial forces. Currently, the design of the wave energy converter to withstand the radial forces is left for a paper to be written.

Theory

The wave energy converter is continuously subjected to loads from the waves, the electromagnetic energy conversion and magnetic circuit, resulting in both radial and axial forces on the hull. During the development of the project, this force has been measured *In-Situ*, and compared to analytical and numerical calculations (Savin et al., 2011; Savin et al., in press). However, every time one or more system-parameters change, the force-distribution in the material changes drastically, making it hard for the mechanical constructors to draw conclusions from earlier prototypes.

The translator moves up and down driven by the motion of the buoy at the surface. As the translator hits the upper end stop, illustrated in Fig 2 and Fig 3, the axial forces, and thereby the stresses in the mechanical structure, reach the maximum value. To perform a correct dimension of the mechanical structure, these forces, stresses and the reaction time of the different components in the damping system, *the excessive overload system* (EOS), have to be calculated.

A solution adopted to damp the forces that can damage the mechanical system is illustrated in Fig. 2 and Fig. 3. The excessive overload system consists in the combination of a spring, the *Upper End Stop*, located in the top of the hull, and a *Rubber Damper*, integrated in the buoy. Every time the buoy is subjected to high axial forces directed upwards, the spring and the rubber damper experience the impact, increasing the deceleration-time of the translator and thereby protecting the hull of the generator from irreversible damages.

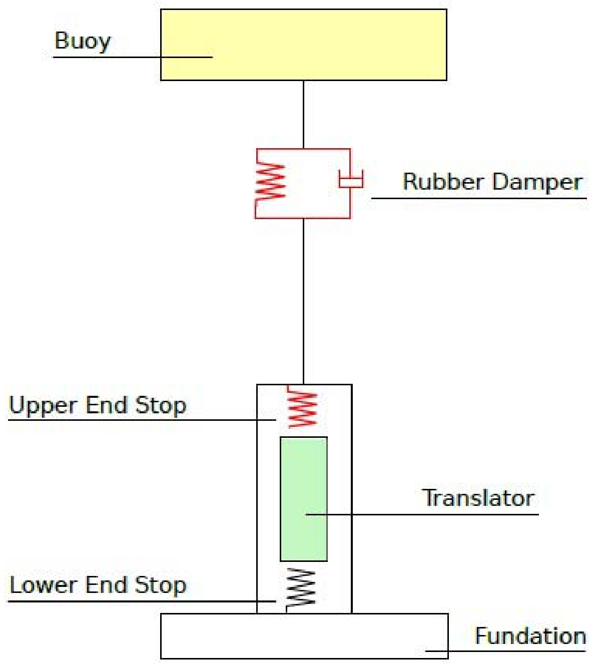


FIG. 2 REPRESENTATION OF THE SPRING AND THE RUBBER DAMPER IN A WAVE ENERGY CONVERTER

The system can be described as a driven damped harmonic oscillation, with the solution:

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F(t) \tag{1}$$

where x is the amplitude of the oscillation, F is the external force, m is the mass compressing the system, k and c are the linear spring constant and the linear damping coefficient, respectively.

The damping coefficient for the spring is assumed to be negligible (Larsson et al, 2006). With the purpose of calculating the time the spring needs to reach the maximum deformation, the following formula has been used:

$$t_s = \frac{\pi}{2}\sqrt{\frac{m}{k}} \tag{2}$$

where the parameter, *m*, represents the mass.

To perform similar calculations on the rubber damper, it is necessary both to calculate the amplitude, *a*, of the oscillation and to estimate the constant k_{sl} for the rubber dumper. The angular frequency, , is obtained by:

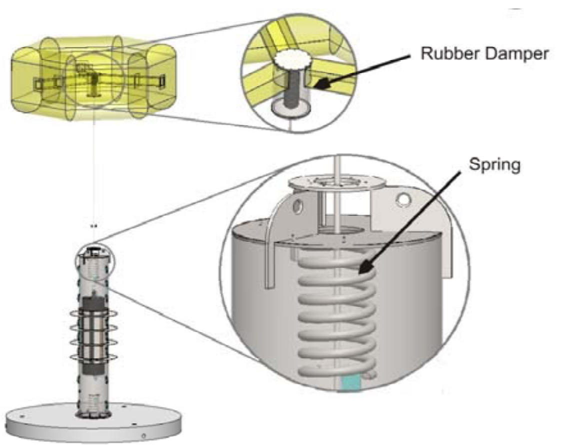
$$\omega = \frac{2\pi}{t} \tag{3}$$

where *t* is the time necessary to complete an oscillation, given by

$$t = \frac{2a}{v} \tag{4}$$

where *v* is the velocity of the oscillation.

FIG. 3 DESIGN OF THE EXCESSIVE OVERLOAD SYSTEM



The natural frequency of the system is dependent on the mass and the constant k_{sl}:

$$\omega_0 = \sqrt{\frac{k_{sl}}{m}} \tag{5}$$

The linear damping coefficient, *c*, has been calculated via the energy-absorption, *E*, of the rubber damper been calculated:

$$c = \frac{E}{\pi a^2 \omega} \tag{6}$$

The critical damping coefficient, c_c, is obtained by the following relation

$$c_c = 2\sqrt{k_{sl} m} \tag{7}$$

and the damping factor that describes the amount of damping in the system is given by the ratio

$$\zeta = \frac{c}{c_c} \tag{8}$$

The phase angle β is defined as

$$\tan\beta = \frac{2\zeta\omega/\omega_0}{1-(\omega/\omega_0)^2} \tag{9}$$

the time delay for the rubber damper is obtained with the formula

$$t_d = \frac{\beta}{2\pi} t \tag{10}$$

and, finally, the reaction time is calculated as (Metev et al., 2005; Zhang et al., 2007; Wegmuller et al., 2000)

$$t_{sl} = \frac{\pi}{2}\sqrt{\frac{m}{k_{sl}}} \tag{11}$$

Design Parameters

The external forces acting on the damping system are in the range 0-300 kN, if both the upwards force produced by the buoy and the downwards force generated by the weight of the translator are taken into account. In this study three different forces have been selected to carry out the calculations: 50, 100 and 300 kN.

The solution proposed by the authors consists of a spring and a rubber damper, with the design parameters summarized in Table 1.

The rubber damper is designed with *EPDM- rubber*, whose main material parameters are presented in Table 2.

TABLE 1 DESIGN PARAMETERS OF THE SYSTEM

Parameter	Value
k [kN/m]	243
$F_{\text{Maximum Force}}$ [kN]	300
k_{sl} [kN/m]	2327
v_{Max} [m/s]	2
E [kNm]	36

TABLE 2 MATERIAL PARAMETERS OF THE RUBBER

Parameter	Value
Yield Strength [MPa]	9.2
Tensile Strength [MPa]	13
Mass Density [g/cm³]	1.02
Poisson's Ration	0.49
Maximum Compression	30 %

TABLE 3 DESIGN DIMENSION

Parameter	Value
Diameter [mm]	240
Hight [mm]	673

The Yield Strength and the maximum amplitude of the force give the area of the circular rubber sheets, whereas the maximum compression of the rubber decides the minimum height of the rubber damper. The dimension of the rubber damper integrated in this axial damper system is presented in Table 3.

As the rubber damper is chosen to be included in the system, the weight of the translator and the electromagnetic damping force constantly compresses the rubber, illustrated in Fig 4, i.e. the rubber is pretensioned.

However, the analytical method, described above fails to take this into account. This assumption is discussed in the next chapter.

Results

The first important result regards the time, t_s, necessary for the spring to reach the maximum deformation. Considering that the spring in the top of the hull is designed for a limit force of 72 kN, as shown in Table 1, this value is included in the calculations. The values obtained from Equation 2 are listed in Table 4.

TABLE 4 MAXIMUM DEFORMATION TIME OF THE SPRING

Force [kN]	50	72	100	300
td [s]	0	0	0	0
ts [s]	0,23	0,27	0,23	0,13

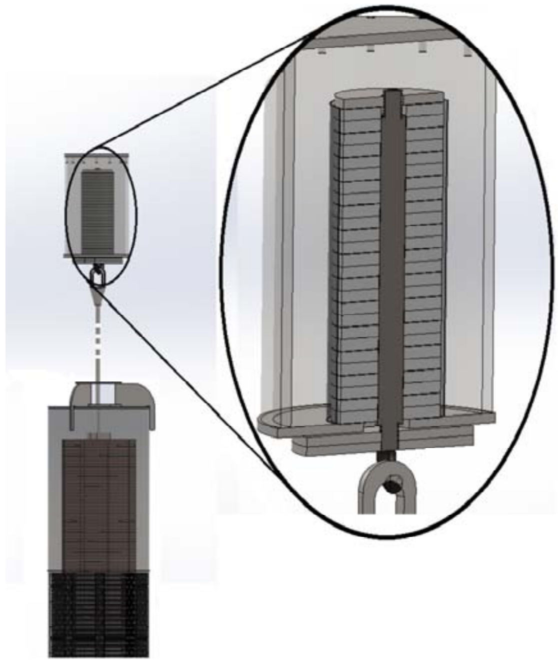


FIG. 4 ILLUSTRATION OF THE CONNECTION BETWEEN THE RUBBER DAMPER AND THE TRANSLATOR

TABLE 5 DELAY TIME AND REACTION TIME FOR THE RUBBER DAMPER

Force [kN]	50	100	300
td [s]	0,00018	0,00019	0,0012
tsl [s]	0,073	0,103	0,178

Then the time delay, t_d, and the reaction time, t_{sl}, for the rubber damper have been calculated and the results are summarized in Table 5. The results are summarized in Fig. 5 and Fig 6 where the time of reaction and the delay for the spring and the rubber damper are indicated by blue symbols. The dashed blue line represents a limit since the maximum load the spring is designed for is 72 kN. The limit fixed by the rubber damper is about 500 kN, hence, it is not reported in the figure. The dashed and dotted lines assume a linear behaviour of the rubber damper and the spring, but this has to be verified with full scale experiments.

Discussion

The results, summarized in Fig 5, present the delay-time of the rubber damper device, i.e. the rubber requires time to react and damp the force, whereas the spring reacts instantly.

By implementing different materials in the excessive overload system and utilizing their different reaction times, the force-distribution in the system can be controlled and directed as expected. To minimize the expensive maintenance of the component placed on

the seabed, it is argued that it is needed to protect the hull of the linear generator from extreme forces.

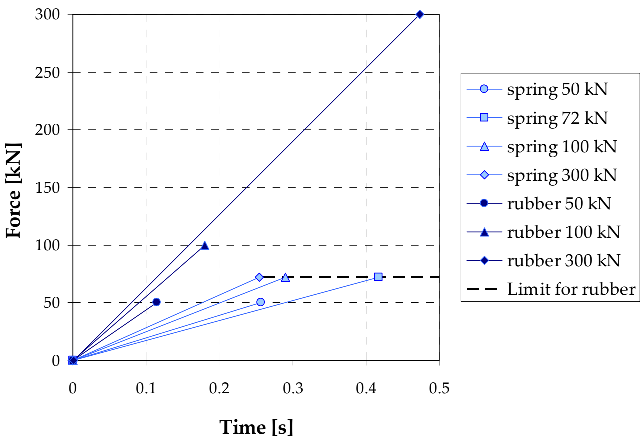


FIG. 5 TIME OF REACTION AND DELAY FOR THE SPRING AND THE RUBBER DAMPER WHEN COMPRESSED BY DIFFERENT FORCES

The direct reaction of the spring results in an instantaneous deceleration of the translator, and thereby it acts as the first step in the damper system. As the system is designed, the compression time of the spring is longer than the time delay of the rubber damper. Therefore the draw-back associated with the time delay for the rubber is not an issue in this particular application. However, if the rubber damper is dimensioned wrongly, the time delay of the reaction can be longer than the compression time of the spring. If then the spring is not capable to decelerate the incoming translator, which hits the hull and exposes the hull of an high force-impact, increasing the risk of a mechanical breakdown. Therefore, taking into account the importance of dimensioning the axial damping force system, the amplitude of the peak forces acting on the buoy as well as the reaction time of the different devices in the system should be in consideration.

The results presented in the previous section show that the time delay associated with each force is lower than the reaction time for the spring. This means that the damping system was correctly dimensioned.

As already mentioned, the pretension of the rubber is not considered in this paper. However, the positive side effects this pretension results in has been argued. Due to the pretension, the rubber is partly compressed, i.e. the compression has already begun. It is believed that this shall reduce the delay time and therefore include the reaction of the rubber damper earlier in the damper system.

When the rubber damper compresses, a relative motion between the buoy and the line appears. This relative motion is considered as a positive side affect, since it makes the buoy free from being further sunk into the water. The lifting force does not increase as it should without this relative motion, i.e. the force amplitude decreased and the lifetime of the hull shall therefore increase.

The nonlinear characteristic behaviour of rubber increases the requirement of experimental verification of the analytical model, which is the next step in the development of the excessive overload system.

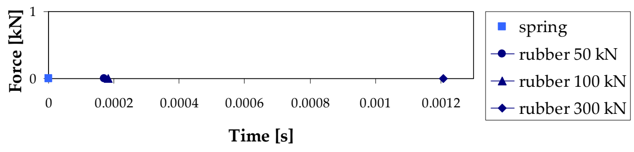


FIG. 5 TO VISUALIZE THE DELAY OF THE SPRING AND THE RUBBER DAMPER, A ZOOMED IN PLOT IS HERE PRESENTED

Conclusion

This paper presents an analytical model of the maximal axial forces in a linear generator, implemented in a wave energy converter. As the development of the complete system goes forward, the economical perspective becomes more and more important. In order to ensure an economically viable alternative to the electric energy conversion, the cost associated to the use of materials has to be reduced while the survivability of the wave power unit has to be increased.

When different materials are integrated in the same system, as the one described above, the reaction time of each material is attached on much importance. The system described in this study results correctly dimensioned, because the compression time of the spring is longer than the time delay of the rubber damper. A reaction time incorrectly calculated can result in a mechanical breakdown. However, by simplifying and modelling each problem with both analytical and numerical models, it is considered that each obstacle can be graded or removed in a correct way, thanks to increasing knowledge and experience.

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